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TITLE: COUNTER-FLOW HEAT EXCHANGER WITH OPTIMAL SECONDARY CROSS-FLOW

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COUNTER-FLOW HEAT EXCHANGER WITH OPTIMAL SECONDARY CROSS-FLOW

BACKGROUND OF INVENTION

Field of Invention

The present invention relates to heat exchangers, such as a counter-flow heat exchanger.

Discussion of Related Art

It is well known in the art that counter-flow heat exchangers can be used particularly in air conditioners of motor vehicles, such as CO_2 air conditioners, in order to provide an accumulator and an internal heat exchanger of the refrigerant circulating system in an integrated arrangement.

Examples of known heat exchangers are disclosed in U.S. Patents Nos. 3,955,375 and 4,895,203 and German Published Patent Application DE 196 35 454 A1. The entire contents of U.S. Patents Nos. 3,955,375 and 4,895,203 are incorporated herein by reference.

In each of the above-identified patents and patent disclosures, a spiral loop is placed between and is in direct contact with the inner and outer walls of the heat exchanger unit. High pressure, high temperature fluid flows within the tubing while low pressure, low temperature fluid flows outside the tubing and between the inner and outer walls of the heat exchanger unit. Such a design can have several drawbacks. First, the fluid outside the spiral tube has areas of higher temperature formed where the spiral tubing contacts the inner and outer walls as shown in the Computational Fluid Dynamics analysis illustrated in FIG. 1. This is believed to be caused by the contact between the spiral tubing and the walls that reduces the velocity of fluid flow and, thus,

reduces heat exchange. Thus, uneven heat exchange occurs along different positions of the spiral tubing.

Another drawback of known spiral heat exchanger units is that physical contact between the spiral tubing and the inner and outer walls contracts the flow path along the spiral tubing and, thus, increases the drop in pressure.

SUMMARY OF THE INVENTION

One aspect of the present invention regards a heat exchanger that includes a first wall that extends along a first direction and defines a first perimeter in a plane that is perpendicular to the first direction and a second wall that defines a second perimeter and is positioned within the first perimeter, wherein the first wall and the second wall are spaced from one another so as to define a volume of space therebetween. A lid is attached to a top portion of the first wall and a top portion of the second wall and a bottom attached to a bottom portion of the first wall and a bottom portion of the second wall. A spiral tubing is positioned within the volume of space, wherein at least a portion of the spiral tubing does not contact either the first wall or the second wall so that a first gap is formed between the first wall and a first portion of the spiral tubing positioned nearest the first wall and a second gap is formed between the second wall and a second portion of the spiral tubing that is positioned nearest the second wall. A first fluid flows within the spiral tubing and a second fluid flows within the first and second gaps.

A second aspect of the present invention regards a method of exchanging heat that includes flowing a high pressure, high temperature fluid within a tubing generally along a first direction and flowing a low pressure, low temperature fluid within a first gap formed between a first wall and a first portion of the tubing positioned nearest the first

wall and a second gap formed between a second wall and a second portion of the tubing that is positioned nearest the second wall.

Each aspect of the present invention provides the advantage of enhancing heat exchange and reducing pressure drop.

The present invention, together with attendant objects and advantages, will be best understood with reference to the detailed description below in connection with the attached drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 schematically shows a side cross-sectional view of a temperature distribution, via Computational Fluid Dynamics analysis, of a heat exchanger without gaps formed between a spiral tubing and walls of the heat exchanger;

FIG. 2 shows a perspective, partially exposed view of a heat exchanger in accordance with the present invention;

FIG. 3 shows a bottom view of the heat exchanger of FIG. 2;

FIG. 4 shows a side cross-sectional view of the heat exchanger of FIG. 2 taken along line 4-4 of FIG. 3;

FIG. 5 shows an enlarged side cross-sectional view of the circled portion A of the heat exchanger of FIG. 4; and

FIG. 6 schematically shows a side cross-sectional view of a temperature distribution, via Computational Fluid Dynamics analysis, of the heat exchanger of FIGS. 2-5.

DETAILED DESCRIPTION OF THE INVENTION

Referring to the drawings, FIGS. 2-5 show an embodiment of a heat exchanger 100 that combines internal heat exchanging with the functions of a refrigerant accumulator, e.g.: preventing liquid from getting to the compressor, refrigerant storage, oil circulation, desiccation, and sensor attachment. As shown in FIG. 2, the heat exchanger 100 can be used particularly for the refrigerant circulation of a motor vehicle air conditioner, in which CO_2, R134a or another conventional refrigerant is used. On the low-pressure side, the collector part of the heat exchanger 100 includes an accumulation chamber 108 that adjoins an evaporator, while, on the high-pressure side, the heat exchanger 100 is situated between a condenser or gas cooler and an expansion valve.

The heat exchanger 100 includes an overall or outer cylinder wall 102 and a high pressure, high temperature tubing 104 laid inside the outer cylinder wall 102 spirally. Consecutive loops of the spiral tubing 104 are separated from one another by approximately 2mm. The spiral tubing 104 is made of a highly thermally conductive material and creates a large heat exchanging area in a confined volume.

In order to improve the heat exchanging effectiveness of the heat exchanger 100, an inner cylinder wall 106 is positioned inside the outer cylinder 102. The inner cylinder wall 106 and outer cylinder wall 102 are co-axial with respect to one another and define in part an accumulation heat transfer chamber 108 therebetween. The heat exchanger 100 includes a cylinder lid 110 that is attached to the upper edges of the inner and outer walls 106 and 102. Similarly, a bottom 112 is attached to the lower edges of the inner and outer walls 106 and 102. The cylinder lid 110 and bottom 112 each form a seal with the inner and outer walls 106 and 102. The inner cylinder wall 106 has an inner

diameter of approximately 60 mm while the outer cylinder wall has an inner diameter of approximately 80 mm. The inner and outer cylinder walls each have a length of approximately 200 mm. The walls 102 and 106 are each made of stainless steel with a thickness of approximately 2 mm.

Note that the inner cylinder wall 106 can be interference fitted, brazed or welded to the cylinder lid 110 on the top and to the outer cylinder wall 102 on the bottom so as to contain the low pressure, low temperature side refrigerant in the heat transfer chamber 108. The inner cylinder wall 106 separates the low pressure, low temperature refrigerant that is in the heat transfer chamber 108 between the inner wall 106 and the outer wall 102 and that flows along the exterior of the spiral tubing 104. The inner cylinder wall 106 also ensures that the spiral tubing 104, used for counter flow heat exchanging, is compactly packed between the two cylinder walls 102, 106.

As shown in FIGS. 4 and 5, the spiral tubing 104 is placed within the heat transfer chamber 108 so that the spiral tubing 104 lies between and does not touch the inner cylinder wall 106 and the outer cylinder wall 102 so that an inner side gap 114 and an outer side gap 116 are formed. The side gaps 114 and 116 are formed along the entire length of the spiral tubing 104. As shown in the cross-sectional view of FIG. 5, for the inner side gap 114 the innermost portion 118 of the spiral tubing 104 is located a distance d1 from the inner cylinder wall 106 that ranges from 0.2 mm to 0.5 mm, preferably 0.3 mm. For the outer side gap 116, the outermost portion 120 of the spiral tubing 104 is located a distance d2 from the outer cylinder wall 102 that ranges from 0.2 mm to 0.5 mm, preferably 0.3 mm. The distances d1 and d2 are chosen so that the inner and outer side gaps 114 and 116 reduce the low side pressure drop without adversely affecting heat exchange performance. The distances d1 and d2 can be

roughly the same. There exist optimal gap widths d1, d2, such as 0.3 mm, for maximizing heat exchanging. The gaps can always be increased from this optimal value to reduce pressure drop further while the heat exchanging may be affected. However, the reduction in pressure drop is faster than the reduction of the heat exchanging performance if the gap widths are very small.

Note that the widths d1, d2 can be varied as a function of the vertical distance z between two adjacent spiral tubing bends so as to achieve an optimal combination of high heat transfer and low pressure drop. The optimal value for the widths d1, d2 as a function of the distance z. In particular, the ratio of the widths d1, d2 to the distance z preferably is approximately 0.1.

Without being held to any particular theory, it is believed that the side gaps 114 and 116 create secondary by-pass flow for the low side refrigerant flow, which primarily follows the trajectory path of the spiral tubing 104 and therefore significantly reduces the speed of the flow along the spiral tubing 104. Such secondary by-pass flow involves having a component of the low side refrigerant to flow between the gaps 114 and 116. This secondary by-pass flow can allow a significant reduction in pressure drop. For example, a considerable reduction of low side total pressure drop (about 50%) without having significant reduction in heat transfer is achieved with widths d1, d2 each having a value of approximately 0.3 mm. In this example, a 2% to 15% increase in heat exchanging is achieved. The percentage of reduction in low side pressure drop along the spiral loop, excluding the effects of the inlet and outlet ports, is much higher.

It is also believed that the side gaps 114 and 116 also allow the low side to flow more evenly at any position around the circumference of the spiral tubing 104. In FIG. 1, heat exchangers without side gaps have higher temperatures at the positions where

the spiral tubing touches the walls, where the flow is very minimal. On the other hand, in the case of the embodiment of the present invention shown in FIGS. 2-5, there is no higher temperature at the position where the spiral tubing is closest to the wall, where there is cross flow from one spiral loop to the other, taking heat from the high side as shown in FIG. 6. This allows for better heat exchanging at any cross section of the tubing. In addition, the high temperature side refrigerant flows a direction opposite to that of the low temperature side refrigerant, which, aside from the primary flow without high-pressure drop has a secondary cross-flow. The benefit of counter flow heat exchanger is retained allowing high effectiveness of the heat exchanger.

Besides providing the advantage of reducing pressure drop without adversely affecting heat exchanging properties, the gaps 114 and 116 allow for more loops of the spiral tubing 104 to be packed per linear length of the walls 102, 106 without incurring a high pressure drop. Accordingly, the gaps 114 and 116 allow for relaxed dimension control, as far as the distance between consecutive loops of the spiral tubing 104 is concerned.

That portion of the heat transfer chamber 108 that is not occupied or displaced by the spiral tubing 104 allows the low pressure, low temperature refrigerant, such as CO_2 or R134a, to flow along the path of the high pressure, high temperature refrigerant within the spiral tubing 104 but in an opposite direction. Such lay-out provides the opportunity to place heat exchanging tubing compactly and allow for counter-flow heat exchanging.

Please note that the inner cylinder wall 106 can be used as an insulator between the liquid refrigerant and the high pressure, high temperature refrigerant. In addition, the

outer cylinder wall 102 can be insulated to prevent the low pressure, low temperature refrigerant from absorbing heat from an engine compartment.

As shown in FIG. 2, the inlet 118 of the spiral tubing 104 extends through the bottom 112 at the step 119 of the outer cylinder wall 102 while the outlet 120 of the spiral tubing 104 extends out of the cylinder lid 110. The inlet 118 and the outlet 120 form seals with the bottom 112 and the cylinder lid 110, respectively. The high pressure, high temperature fluid flows from the inlet 118, through the spiral portion of the tubing 104 and out of the outlet 120.

The low pressure, low temperature fluid flows in a direction generally opposite than that of the high pressure, high temperature fluid. As shown in FIG. 4, an inlet 122 for the low pressure, low temperature fluid is formed in the cylinder lid 110. An outlet 124 for the low pressure, low temperature fluid is formed in the bottom 112.

In operation, a high pressure, high temperature fluid, such as CO_2 or R134a, from a gas cooler 126 or condenser 128 flow along the high pressure, high temperature inlet 118, runs through the spiral loop portion of the spiral tubing 104 and exits out of the high pressure, high temperature outlet 120 which is connected to an expansion valve and an inlet of an evaporator 130. While the high pressure, high temperature fluid is flowing from inlet 118 to outlet 120, the low pressure, low temperature fluid from the outlet of the evaporator 130 flows along the low pressure inlet 122 and into the heat transfer chamber 108 between the walls 102, 106 and the spiral tubing 104. The fluid portion of the low pressure, low temperature fluid mostly deposits at the bottom of the accumulator 100 while the vapor portion of the low pressure, low temperature fluid exits the accumulator 100 via the top portion 132 of a J-tube 134. The function of the J-tube

134 is to pick up oil and to prevent liquid refrigerant from reaching the compressor section.

As shown in FIG. 4, an orifice-filter 135 is attached to the J-tube 134. The orifice filter 135 picks a certain amount of liquid from the accumulator 100.

One or more oil suction bores (not shown) are provided in the lower area of the accumulator 100. The oil suction bores are dimensioned such that, as a function of the suction effect, the more viscous fluid is sucked to a certain desired extent out of the accumulator 100. In the accumulator 100, the intermediately stored refrigerant is situated in the lower area above the settled oil in the liquid state and in the upper area in the gaseous state.

As a result of the suction effect of the compressor 125, the vapor within the J-tube 134 flows into the space 136 defined by the inner cylinder wall 106 and flows out of the space 136 via an outlet 138 of the J-tube 134 that is located near the top of the inner cylinder wall 106. The outlet 138 is in fluid communication with the heat transfer chamber 108 that is defined as the space between walls 102 and 106 not displaced by the spiral tubing 104. The vapor flows downward in the heat transfer chamber 108 between the coils of the spiral tubing 104 and the spaces between the spiral tubing and inner and outer walls and enters into a withdrawal space, again entraining by way of the suction bores and exits out of the low pressure, low temperature outlet 124 formed in the bottom 112. The outlet 124 is connected via a tube and hose to the compressor 125.

Note that a zero leak joint is only needed on the joints to contain the low pressure, low temperature side refrigerant. Such a zero leak joint includes the lid-outer cylinder joint, which also requires the highest joint strength to contain the pressure, the

inlet and outlet fittings. The other internal joints are more tolerable about sealing; brazing satisfies requirements but is not always required.

It is possible to create surface irregularities either on both of the contacting cylinder walls, or on the spiral tubing 104. For example, the inner and outer cylinder walls can have such a profile. Then, the cylinder walls and the spiral tubing 104 are put into position, and the inner cylinder wall 106 is expanded to have the spiral tubing 104 tightly in touch on both sides, with the irregularities creating the optimal side gaps between the spiral tubing 104 and the inner and outer cylinder walls.

In another variation of the present invention, the single spiral tubing 104 of FIGS. 2, 4 and 5 is replaced by a double spiral tubing. Thus, more heat exchanging tubing is packed in heat exchanger 100.

Another embodiment of a heat exchanger according to the present invention regards varying the flow pattern of the heat exchanger 100 of FIGS. 2-5. This is accomplished by altering the heat exchanger 100 of FIGS. 2-5 so that the spiral tubing 104 has an inlet formed at the top cylinder lid 110 and an outlet formed at the bottom 112. A temperature and/or pressure transducer that is required at the heat transfer chamber 108 can be attached at the top of the heat exchanger 100. In this embodiment, the high pressure, high temperature fluid from the gas cooler or condenser flows along the high pressure, high temperature side inlet and runs through the spiral tubing 104 from the top to the bottom of the heat exchanger 100 and exits out of the high pressure, high temperature outlet connected to the evaporator. While the high pressure, high temperature flows downward, the low pressure, low temperature fluid flows upward by entering through the bottom 112 via of the inner cylinder wall 106 and into the heat transfer chamber 108. The fluid portion of the low pressure, low

temperature fluid mostly deposits at the bottom of the accumulator while the vapor portion of the low pressure, low temperature fluid exits the accumulator via the top portion 132 of a J-tube 134. An orifice-filter 135 is attached to the J-tube 134. One or more oil suction bores are provided in the lower area of the accumulator. The structures and functions of the J-tube 134 and suction bores are the same as those used in the heat exchanger 100. In this embodiment, the orifice filter 135 performs an additional function besides picking up oil in that it also picks a certain amount of liquid from the accumulator.

The vapor portion of the low pressure, low temperature fluid then flows between the loops of the spiral tubing 104 and the inner and outer side gaps 114 and 116, respectively. The vapor portion exits out of the low pressure, low temperature outlet formed in the cylinder lid 110. The outlet is connected via a tube and hose to the compressor.

The foregoing description is provided to illustrate the invention, and is not to be construed as a limitation. Numerous additions, substitutions and other changes can be made to the invention without departing from its scope as set forth in the appended claims.